

Thermodynamic analysis of the heat regenerative cycle in porous medium engine

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ABSTRACT

The advantages of homogeneous combustion in internal combustion engines are well known all over the world. Recent years, porous medium (PM) engine has been proposed as a new type engine based on the technique of combustion in porous medium, which can fulfil all requirements to perform homogeneous combustion. In this paper, working processes of a PM engine are briefly introduced and an ideal thermodynamic model of the PM heat regeneration cycle in PM engine is developed. An expression for the relation between net work output and thermal efficiency is derived for the cycle. In order to evaluate of the cycle, the influences of the expansion ratio, initial temperature and limited temperature on the net work and efficiency are discussed, and the availability terms of the cycle are analyzed. Comparing the PM heat regenerative cycle of the PM engine against Otto cycle and Diesel cycle shows that PM heat regenerative cycle can improve net work output greatly with little drop of efficiency. The aim of this paper is to predict the thermodynamic performance of PM heat regeneration cycle and provide a guide to further investigations of the PM engine.

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1. Introduction

The development of internal combustion (IC) engines has achieved a high level of success in the last century. Recent research of IC engines has been motivated by the desire to preserve a clean environment and to reduce energy consumption. Reducing exhaust emissions and increasing the fuel efficiency of internal combustion engines are of global importance. Currently, homogenous charge compression ignition (HCCI) engines are being actively investigated worldwide as they can achieve efficiencies close to that of diesel engines and, at the same time, with low levels of oxides of nitrogen (NO_x) and particulate matter emissions [1]. Despite many advantageous features of HCCI combustion, it still fronts some challenges including higher hydrocarbon (HC) and carbon monoxide (CO) emissions, the control of ignition timing and combustion rate and expanding operating range, especially if variable engine operational conditions are considered [2]. In this respect, the new concept of the so called porous medium (PM) engine, based on the regenerative or super-adiabatic combustion in porous medium, offers new ways in dealing with these problems. Recently, the PM engine has received more and more attention from numerous researchers because of its potential for producing homogeneous mixtures and reducing NO_x and soot emissions [3,4].

The technique of premixed combustion within porous media has been studied intensively and applied to steady combustion with great successes over the past decades [5,6]. Consequently, the technique has been then extended from gaseous to liquid fuels and from steady to unsteady combustion. Liquid vaporizing combustion in porous medium burner has fine flame stability and characteristic of low emission, which has been proved by experimental and numerical research [7–10]. On this basis, the new concept of controllable combustion in porous media for internal combustion engine was suggested and developed. Durst and Weclas [11] proposed two designs of the PM engine, in one of which a porous medium combustion chamber is mounted in the engine cylinder head, fuel is injected into the porous medium volume, and consequently, all combustion events, i.e. fuel vaporization, fuel–air mixture formation and homogenization, internal heat recuperation, as well as combustion reaction occur inside the porous medium. In order to prove the feasibility of PM engine, they modified a single-cylinder, air cooled diesel engine to incorporate a porous medium reactor in the cylinder head and operated it as a PM engine (Fig. 2). The engine was operated for several hours without any damage to the PM material or showing any uncontrolled operations, unfortunately, the detailed results was not published in any literature. Hanamura [12] designed a reciprocating heat engine, which is similar to a Stirling engine with super-adiabatic combustion in porous media. One-dimensional numerical simulations showed that the thermal efficiency of the engine reached to 57.5% under even very low compression ratios between 2 and 3, which are much lower

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than those of conventional Otto and Diesel cycles. Weclas [13] proposed a strategy for development of intelligent combustion systems for IC engines, whose essence was a new concept for mixture formation (MDI – mixture direct injection) and homogeneous combustion based on the Porous Medium technology. Macek [14] analyzed the possibilities of homogeneous combustion achieved by porous medium with limited temperature, and found thermodynamic limits of a new cycle with PM combustion using high flame stability and fast burning at comparatively low temperatures and the potential of internal heat regeneration.

The objective of this paper is to investigate the PM heat regenerative cycle in a PM engine and evaluate its thermodynamic performance. The engine is derived from one of the designs of Durst [11] and the analysis is based on general idealized cycle model. An ideal thermodynamic model for the cycle of PM engine is presented to evaluate effects of various working parameters on the thermodynamic performance of the PM engine. The ideal model of the PM heat regenerative cycle is according to the concept of PM engine advanced by Durst and Weclas [11,13] and has been validated by comparison with the analysis Macek [14,15].

2. Principle and theoretical model of the PM engine

2.1. Principle of the PM engine

The PM engine is here defined as an internal combustion engine with a highly porous medium chamber mounted on the cylinder head (Fig. 1). The PM chamber is thermally isolated from the head walls and equipped with a valve permitting a periodic contact between the PM-chamber and the cylinder volume.

Fig. 1 shows the complete working cycle of the PM engine advanced by Durst [11], and the view of the PM engine head with SiC reactor is shown in Fig. 2. Near the TDC the intake valve opens and fresh air flows into the cylinder, the valve of the PM-chamber keeps closed and the PM volume containing fuel vapor has no contact with the fresh air during intake and compression strokes. At the end of compression the valve of the PM-chamber opens and the compressed air flows from the cylinder into the hot PM-volume, generating a highly turbulent flow condition. As a result, very fast mixing of the vaporized fuel and compressed air occurs in the PM-chamber. The mixture receives enough heat from the hot PM and is then auto-ignited in the entire PM-volume. The resulting flame

passes through the porous matrix with a high velocity and a complete combustion takes place in the PM-chamber. During the combustion process, the heat being released is absorbed by both working gas and PM-volume. The valve of the PM-chamber remains open until the end of the expansion stroke and fuel is then injected in the PM-volume after closing the valve. The process of vaporization in PM-volume continues throughout exhaust, intake and compression strokes. The large specific PM surface area and excellent heat transfer in the PM-volume as well as the long time available for fuel vaporization make the liquid fuel vaporization very fast and complete. The cylinder and the PM-chamber keep separated during the exhaust processes, which is same with conventional DI engines.

The valve of the PM-chamber switches on and off one time each cycle and the PM-chamber periodically contacts with working fluid. The PM absorbs heat during combustion period and releases it at the end of the next compression process.

The formation of homogeneous mixture and 3D-thermal self-ignition in the PM-volume create a realizable condition for homogeneous combustion, which is almost independent of the engine load. Furthermore, the heat recuperation in PM may be used for heating up the compressed air and controlling the combustion temperature level.

2.2. Idealized thermodynamic modeling of porous medium engine

To conduct an ideal cycle analysis of the PM engine, three essential assumptions were adopted in this study, which have been used commonly in researches on conventional engines and on the homogeneous combustion within porous medium [11].

- (1) The heat capacity of porous medium is much larger than that of gas, thus the temperature of porous medium can be regarded as constant and not affected by the heat exchange between the porous medium and the working gas.
- (2) Heat losses via the piston, cylinder wall and PM-chamber are neglected. The compression and expansion processes realized were considered adiabatic.
- (3) There are no time elapses during the thermal coupling between the PM-volume and the cylinder. This means the heat transfer between porous medium and the work gas is completed instantaneously.

Under these assumptions, an idealized thermodynamic cycle with PM heat regeneration in the PM engine can be described with Fig. 3 which presents P-V and T-S diagrams for the heat regeneration cycle of the PM engine (1–2–3–3'–4–1) and for a conventional Otto cycle (1–2–3–4'–1) and Diesel cycle (1–2–3'–4'–1) under the same temperature limits, respectively. For the PM heat regeneration cycle, line 1–2 is isentropic compression process, which is same as in the Otto cycle and Diesel cycle in Fig. 3. Heat regeneration process begins at the end of compression when the PM couples with the cylinder and the whole process proceeds so fast that it can be modeled as a constant volume process (2–3). The mixture filled in the PM-volume obtains sufficient heat from the porous medium during the heat regenerative process and, consequently, is self-ignited at TDC. Since the porous medium has very large heat capacity, the combustion in PM scarcely increases the temperature of the cycle. Hence the temperature of working fluid remains also nearly constant during the combustion in porous medium. Thus the combustion can be considered as an isothermal heating process (3–3'). The expansion process is an isentropic one (3'–4), which is identical with the Diesel cycle; and the exhaust process (4–1) is an isochoric one, which is same with the Otto cycle (4'–1).

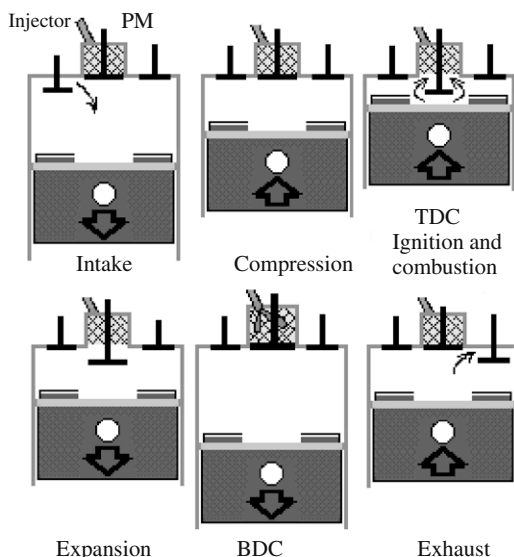


Fig. 1. Principle of the PM engine proposed by Durst.

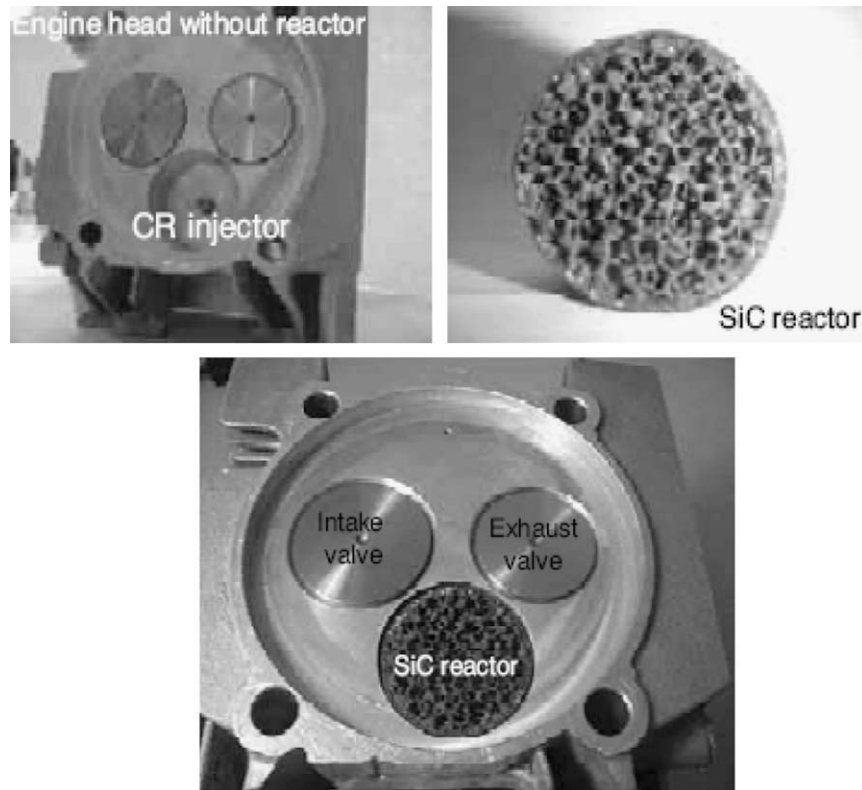


Fig. 2. View of the PM engine head and SiC reactor.

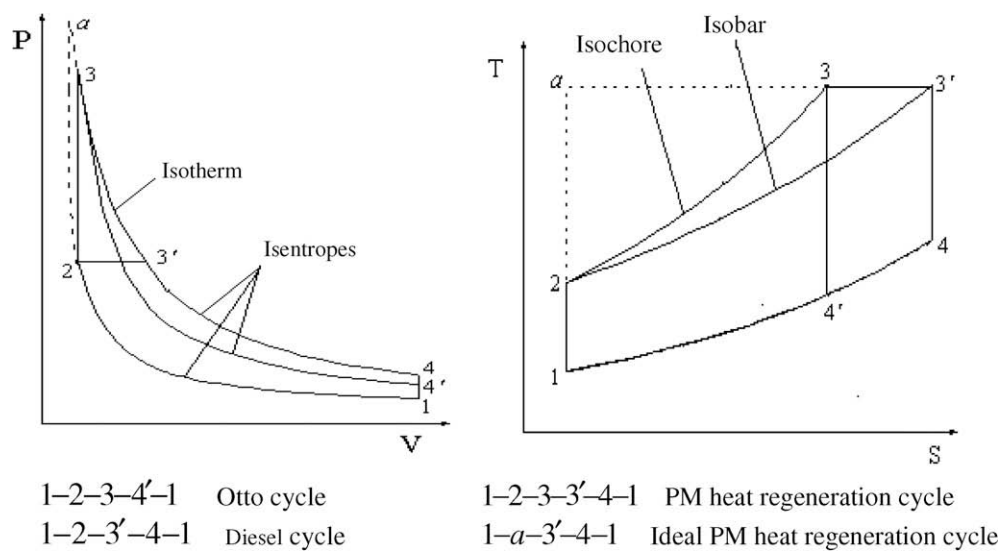


Fig. 3. Comparison of Otto, Diesel and PM heat regeneration cycle.

The main difference between the PM heat regeneration cycle and an Otto cycle is that in the PM heat regeneration cycle the heat absorbed by working fluid is from the porous medium volume during the process 2–3' as well as from combustion reaction during the process 3–3', while in the Otto cycle it is only from combustion during process 2–3. The original isochoric heating process (2–3) in the Otto cycle is replaced now by a combination of an isochoric heat regeneration process (2–3) and an isothermal heating process (3–3'). The net work done in the Otto cycle is characterized with

the area 1–2–3–4'–1. In the case of the PM heat regenerative cycle, the work is represented by the area 1–2–3–3'–4–1, which has a potential work gain corresponding to the area 3–3'–4–4'–3.

We use ratios between state parameters during compression and heating processes as essential variables to describe the working cycle. The compression ratio ε and the expansion ratio ρ are defined as $\varepsilon = v_1/v_2 = v_4/v_3$ and $\rho = v'_3/v_3$, respectively, where v is the volume of the working fluid in the cylinder. The pressure increase ratio is introduced as $\lambda = p_3/p_2$ where p is the pressure of

the working fluid in the cylinder. With given initial values of pressure p_1 and temperature T_1 , the PM heat regenerative cycle in PM engine can be determined by ε , λ and ρ .

Based on the model described above, relationships of characteristic temperatures can be expressed as follows:

$$T_2 = T_1 \varepsilon^{k-1}; \quad T_3 = \lambda \varepsilon^{k-1} T_1; \quad T'_3 = T_3; \quad T_4 = \lambda \rho^{k-1} T_1 = T_3 (\rho/\varepsilon)^{k-1} \quad (1)$$

where T_i is the working fluid temperature at the state (i).

The heat added per unit mass of working fluid for the PM heat regenerative cycle is

$$Q_{in} = Q_{2-3} + Q_{3-3'} = C_v(T_3 - T_2) + RT_3 \ln \left(\frac{v'_3}{v_3} \right) \quad (2)$$

where C_v is the specific heat at constant volume, R is the gas constant of the working fluid, $R = C_p - C_v = (k-1)C_v$, C_p is the specific heat at constant pressure and k is the ratio of specific heats, $k = C_p/C_v$. C_v , R and k are assumed to be constants. The net work output per cycle is

$$W = W_{2-3} + W_{3-3'} + W_{4-1} \\ = C_v(T_3 - T_2) + RT_3 \ln(V_{3'}/V_3) + C_v(T_1 - T_4) \quad (3)$$

By substituting the temperature relationships (1) into Eqs. (2) and (3), the heat added and the net work output per cycle are transformed into

$$Q_{in} = C_v[T_3 + (k-1)T_3 \ln \rho - T_1 \varepsilon^{k-1}] \quad (4)$$

$$W = C_v\{(1 - \varepsilon^{k-1})T_1 + [(k-1) \ln \rho - (\rho/\varepsilon)^{k-1} + 1] \cdot T_3\} \quad (5)$$

The cycle efficiency for the PM heat regenerative cycle 1–2–3–3'–4–1 is

$$\eta = 1 - \frac{T_3(\rho/\varepsilon)^{k-1} - T_1}{T_3 - T_1 \varepsilon^{k-1} + (k-1)T_3 \ln \rho} \quad (6)$$

With same method, the net work output per cycle and the cycle efficiency for Diesel cycle can be expressed as

$$W = C_v\{T_1 - kT_1 \varepsilon^{k-1} + T_3[k - (T_3/T_1)^{k-1} \varepsilon^{k(1-k)}]\} \quad (7)$$

$$\eta = 1 - \left[\left(\frac{T_3}{T_1 \varepsilon^{k-1}} \right)^k - 1 \right] / \left(k \frac{T_3}{T_1} - k \varepsilon^{k-1} \right) \quad (8)$$

In the model analysis mentioned above, the cycle of PM engine is identical with the analytical model of Macek [14] and the analysis is based on general idealized cycle model of classical thermodynamics. However, there are still some questions in the model: For a practical cycle, the necessary requests for the realization of the auto ignition in the PM reactor such as the minimum of compression ratios and initial temperature are not discussed. Besides, the temperature of porous medium is mainly decided by the heat transfer in porous medium, which is effected by the volume heat transfer coefficient and time elapsed. Therefore, the temperature of porous medium will change even for a large heat capacity, i.e. the heating process is merely approximated to isothermal process. In the ideal model, the irreversible changes are not taken into account, e.g. heat loss between the working fluid and cylinder wall, and the friction loss of the piston. Moreover, heat transfer from porous medium to gas especially during early compression and late expansion or even exhaust will cause the loss of energy, which should be taken into account by experiments or more detailed cycle simulation.

2.3. Limit conditions for the PM heat regenerative cycle

There are two extreme conditions for the PM heat regenerative cycle:

1. As shown in Fig. 3, When $\rho = 1$, point 3'-shifts to Point 3 along the isothermal line and the PM heat regenerative cycle switches to the Otto cycle. Substituting $\rho = 1$ into Eqs. (3) and (4) yields the net work output and efficiency of Otto cycle, which have already been proved validated

$$W = c_v[(1 - \varepsilon^{k-1})T_3 + T_1(1 - \varepsilon^{1-k})] \quad (9)$$

$$\eta = 1 - \varepsilon^{1-k} \quad (10)$$

2. With the increasing of compression ratio, as in Fig. 3, point 2 can reach to isothermal line (3–3') along isentropic line (1–2). As $\lambda = 1$, i.e. the isentropic line extend till to point a, the PM heat regenerative cycle switches to a optimal performance under same limit temperature, which is defined as ideal PM heat regenerative cycle (1–a–3'–4–1 in Fig. 3) in this paper. The limit temperature can be expressed as $T_3 = T_2 = T_1 \varepsilon^{k-1}$.

According to the property of isentropic processes, it can be given that $T_a = T_3 = T_1 \varepsilon^{k-1}$. Substituting $\varepsilon = (T_3/T_1)^{1/(k-1)}$ into Eqs. (5) and (6) yields the net work output and efficiency of ideal PM heat regenerative cycle:

$$W = c_v[(1 - \rho^{k-1})T_1 + (k-1)T_3 \ln \rho] \quad (11)$$

$$\eta = 1 - \frac{(\rho^{k-1} - 1)T_1}{(k-1)T_3 \ln \rho} \quad (12)$$

Eqs. (11) and (12) show that the critical parameters for ideal PM heat regenerative cycle are the expansion ratio, the initial temperature and the limit temperature. For example, if the initial temperature $T_1 = 300$ K, the maximum temperature $T_3 = 1800$ K, the expansion ratio $\rho = 1.5$, and $k = 1.4$ the efficiency of the ideal PM heat regenerative cycle can be $\eta = 0.7889$. However, this optimal condition can not be realize actually, because a very large compression ratio and infinite capacitance of PM are required, as in this example, the corresponding compression ratio must be 60, it is hardly to reach. Thus the analysis of the ideal PM heat regenerative cycle is not discussed detailed in this paper.

2.4. Availability analysis for the PM heat regenerative cycle

The availability of a system in a given state is defined as the maximum reversible work that can be produced through interaction of the system with its surroundings, as it reaches thermal, mechanical and chemical equilibrium [16].

In order to further improve thermal efficiency of PM engine, analysis of availability in the PM heat regenerative cycle is necessary. Consequently, entropy is used as a state parameter to describe availability and the value of entropy at initial state is taken appointed as zero. For the PM heat regenerative cycle, we have the value of entropy for each state

$$s_1 = 0; \quad s_2 = 0; \quad s_3 = c_v \ln(T_3/T_2) = c_v \ln(p_3/p_2) = c_v \ln \lambda;$$

$$s_{3'} = s_3 + R \ln(V_{3'}/V_3) = c_v \ln \lambda + (k-1)c_v \ln \rho;$$

$$s_4 = s_{3'} = c_v \ln \lambda + (k-1)c_v \ln \rho;$$

The availability entering into the cycle is

$$E_{in} = Q_{2-3} + Q_{3-3'} - T_1(\Delta S_{2-3} + \Delta S_{3-3'}) \quad (13)$$

The availability removing from the cycle is

$$E_{out} = Q_{4-1} - T_1 \Delta S_{4-1} \quad (14)$$

The availability converting into net work output per cycle can be expressed as the algebraical summation of Eqs. (13) and (14):

$$E = E_{in} + E_{out}$$

$$= Q_{2-3} + Q_{3-3'} + Q_{4-1} - T_1(\Delta S_{2-3} + \Delta S_{3-3'} + \Delta S_{4-1}) \quad (15)$$

Substituting the values of entropy at each state into Eq. (15), it can be given:

$$E = c_v(T_3 - T_2) + RT_3 \ln \lambda + c_v(T_1 - T_4) \quad (16)$$

With same method, the availability destruction for each processes are expressed as followed:

The availability destruction for isochoric heat regenerative process

$$I_{2-3} = T_1 \Delta S_{2-3} = T_1 c_v \ln \lambda \quad (17)$$

The availability destruction for an isothermal heating process

$$I_{3-3'} = T_1 \Delta S_{3-3'} = T_1 (k-1) c_v \ln \rho \quad (18)$$

The availability destruction for isochoric heat released process is

$$I_{4-1} = -(Q_{4-1} - T_1 \Delta S_{4-1})$$

$$= c_v T_4 - T_1 c_v [1 + \ln \lambda + (k-1) \ln \rho] \quad (19)$$

The total availability destruction is

$$I_{total} = I_{2-3} + I_{3-3'} + I_{4-1} = c_v (T_4 - T_1) \quad (20)$$

The magnitude of contribution of each process to the total availability destruction are calculated by Eqs. (17)–(20).

3. Numerical examples

Based on the formulations presented in the previous section, a parametric study was conducted to analyze effects of ρ , T_1 and T_3 on the characteristics of the net work output versus efficiency for PM heat regenerative cycle with ideal thermodynamic model. Range of the initial temperature is $T_1 = 300$ – 350 K, which is adjacent to room temperature. The expansion ratio (ρ) in the PM heat regenerative cycle is limited by the volume of the PM, we take its value as in the range of 1–2.4, which is less than that of the Dual cycle. The limit temperature (T_3) range is 1600–2000 K according to conventional engine. A standard value of 1.4 for k and a value of 0.7165 kJ/(kgK) for C_v are used in the calculations. Results of the calculations for above parameters ranges are shown in the following.

Figs. 4 and 5 indicate respectively the net work output and the thermal efficiency versus the compression ratio ε for the PM heat regenerative cycle without regard to the effect of irreversibility. In the figures, results for the conventional Otto cycle and Diesel cycle are also shown for a comparison. A same initial temperature of 300 K and the maximum temperature of 1800 K are assumed for all

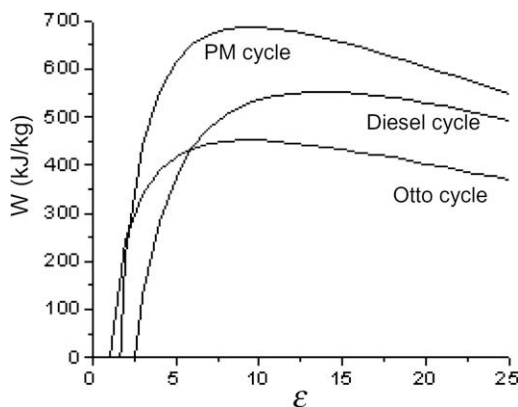


Fig. 4. Comparison of the net work output for Otto, Diesel and PM heat regeneration cycle.

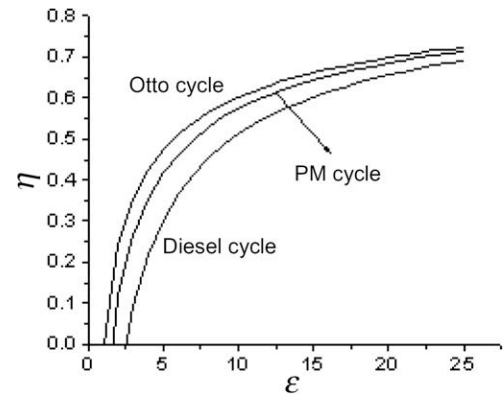


Fig. 5. Comparison of efficiency for Otto, Diesel and PM heat regeneration cycle.

cycles. The expansion ratio of the PM heat regenerative cycle is given as 1.5. The curves in Figs. 4 and 5 show clearly that the thermal efficiency rises continuously with the compression ratio while there exists a maximum for the net work output. The maximum net work output for the PM heat regenerative cycle is at ε about 10, and the corresponding thermal efficiency is about 0.6. As shown in Fig. 5, the thermal efficiency of the PM heat regenerative cycle is slightly lower than that of the Otto cycle, whereas much higher than that of Diesel cycle for the same compression ratio. On the other hand, Fig. 4 shows the net work output of the PM heat regenerative cycle is remarkably larger than that of Otto cycle for compression ratios larger than about 2. It should be noted that, for an actual engine, the compression ratio must exceeds certain value to ensure the realization of the actual cycle, therefore, the net work output of the PM heat regenerative cycle must be larger than that of actual Otto and Diesel cycle. That means the PM heat regeneration cycle can provide significantly more net work output at little expense of thermal efficiency.

Fig. 6 shows the influences of the expansion ratio (ρ) on the net work output versus the efficiency for the PM heat regenerative cycle at a condition of $T_1 = 300$ K and $T_3 = 1800$ K. It is shown that there exists a maximum net work output for constant expansion ratio, with the increasing of the expansion ratio, the maximum net work output increases greatly and the thermal efficiency corresponding to the maximum net work output increases also. When the expansion ratio equals 1 the cycle becomes an Otto cycle, whose net work output curve is much lower than others.

Fig. 7 shows the effects of the initial temperature T_1 on the net work output versus the thermal efficiency for the PM heat regenerative cycle at a condition of $\rho = 2.0$ and $T_3 = 1800$ K. As shown in Fig. 7, the maximum net work output and the corresponding ther-

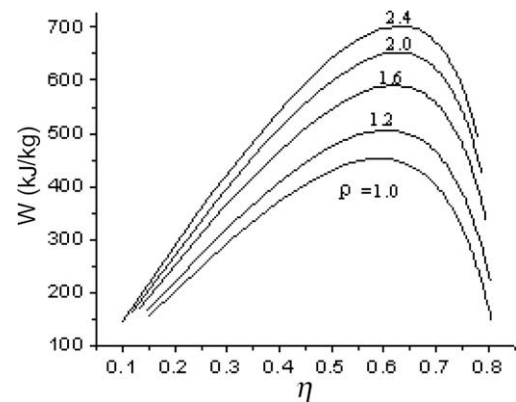


Fig. 6. Influence of ρ on the net work output versus efficiency characteristic.

mal efficiency both decrease with the increasing of the initial temperature, however, the change is not very evident.

Fig. 8 shows the effects of the maximum temperature T_3 on the net work output versus the thermal efficiency for the PM heat regenerative cycle at a condition of $\rho = 2.0$ and $T_1 = 300$ K. It is shown that there exists a maximum net work output for constant maximum temperature, with the increasing of the maximum temperature, the maximum net work output increases evidently and the thermal efficiency corresponding to the maximum net work output increases also.

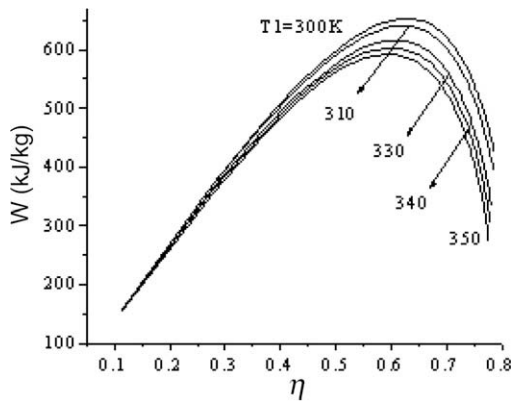


Fig. 7. Influence of T_1 on the net work output versus efficiency characteristic.

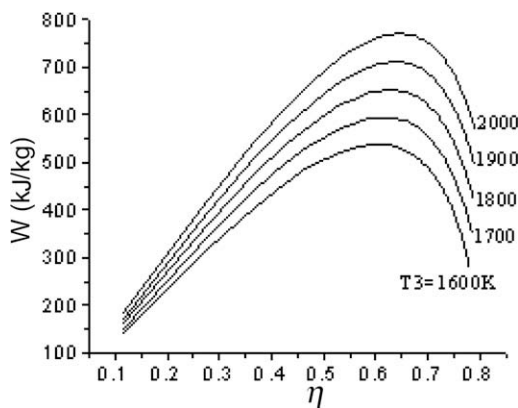


Fig. 8. Influence of T_3 on the net work output versus efficiency characteristic.

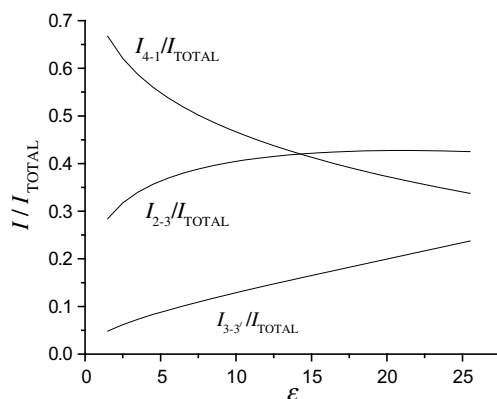


Fig. 9. The magnitude of contribution of each process to the total availability destruction.

Fig. 9 concentrates on the magnitude of contribution of each process to the total availability destruction. The proportion of the availability destruction of isochoric heat regenerative process (2–3), the isothermal heating process (3–3') and the isochoric heat released process (4–1) to the total availability destruction are provided, respectively. As shown in Fig. 9, the availability destruction of the isothermal heating process is the least one, which is always less than 20%. That means the special isothermal heat addition process in the PM heat regenerative cycle can play a positive role in reducing the availability destruction of the cycle. With the increasing of compression ratio, the availability destruction of the isochoric heat released process decreases, on the contrast, those of the isochoric heat regenerative process and isothermal heating process increase. The availability destruction is mainly concentrated on the isochoric heat released process as compression ratio less than about 15.

4. Conclusions

This paper demonstrates an ideal model of the PM heat regenerative cycle in a new type of PM engine. The novel feature of the PM heat regenerative cycle is the heat feedback and an isothermal heat addition process are realized by using the porous medium as a heat recuperator. A relationship between the net work output and thermal efficiency is derived, and two limit conditions of the PM heat regenerative cycle are analyzed. The availability destructions of each process within a working cycle are analyzed by method of entropy analysis and the main source of energy loss in the engine is pointed out. Numerical computations show that the PM heat regenerative cycle can provide much larger net work output than that of an Otto cycle at a little expense of thermal efficiency, and the effects of expansion ratio and limited temperature on the net work output are evidently. It is worthwhile to point out that the special isothermal heat addition process in the PM heat regenerative cycle can play a positive role in reducing the availability destruction of the cycle. The results obtained here could provide significant guidance for the performance evaluation and improvement of practical PM engines.

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References

- [1] Yap D, Megaritis A, Wyszynski ML. An investigation into bioethanol homogeneous charge compression ignition (HCCI) engine operation with residual gas trapping. *Energy Fuels* 2004;18(5):1315–23.
- [2] Zhao H, Peng Z, Ladommatos N. Understanding of controlled auto-ignition combustion in a four-stroke gasoline engine. *Inst Mech Eng D J Automot Eng* 2003;215(4):1297–310.
- [3] XIE MZ. New type of internal combustion engine – superadiabatic engine based on the porous-medium combustion technique. *J Therm Sci Technol* 2003;3(2):189–94.
- [4] Chan WP, Massoud K. Evaporation-combustion affected by in-cylinder reciprocating porous regenerator. *ASME* 2002;124(6):184–94.
- [5] Kakutkina NA. Some stability aspects of gas combustion in porous media combustion. *Explosion Shock Waves* 2005;41(4):395–404.
- [6] Mishra SC, Steven M. Heat transfer analysis of a two-dimensional rectangular porous radiant burner. *Int Commun Heat Mass Transf* 2006;33(2):467–74.
- [7] Echigo R, Yoshida H. Mathematical model of self-sustaining combustion in inert porous medium with phase change under complex heat transfer. *Int J Heat Mass Transf* 1998;41(1):117–26.
- [8] Sumrerng J, Narongsak W. The combustion of liquid fuels using a porous medium. *Exp Therm Fluid Sci* 2002;26(3):15–23.
- [9] Takuya Fuse, Yugo Araki. Combustion characteristics in oil-vaporizing sustained by radiant heat reflux enhanced with higher porous ceramics. *Fuel* 2003;82(5):1411–7.
- [10] Tarun KK. Modeling of trickle flow liquid fuel combustion in inert porous medium. *Int J Heat Mass Transf* 2006;49(1):975–83.

- [11] Durst F, Weclas M. A new concept of I.C. engine with homogeneous combustion in a porous medium. COMODIA 2001:467–72.
- [12] Hanamura K. A feasibility study of reciprocating-flow super-adiabatic combustion engine. JSME Int J 2003;46(4):579–85.
- [13] Weclas M. Strategy for intelligent internal combustion engine with homogeneous combustion in cylinder. ISSN 1616-0762 Sonderdruck Schriftenreihe der Georg-Simon-Ohm- Fachhochschule Nürnberg Nr; 2004.
- [14] Macek J, Polášek M. Via homogeneous combustion to low NO_x emission. In: Proceedings of EAEC congress – CD-ROM. Bratislava: SAITS; 2001. 1:1–10.
- [15] Miloš Polášek, Jan Macek, Homogenization of combustion in cylinder of CI engine using porous medium, SAE Paper 2003-01-1085; 2003.
- [16] Rakopoulos CD, Giakoumis EG. Availability analysis of a turbocharged diesel engine operating under transient load conditions. Energy 2004;29(2): 1085–104.